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Rotor/Stator Unsteady Pressure Interaction

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Abstract

This paper describes an investigation of rotor/stator interaction in centrifugal pumps with radial diffusers. Steady and unsteady diffuser vane pressure measurements were made for two impellers, one half of the double suction pump of the High Pressure Oxygen Turbopump (HPOTP) of the Space Shuttle Main Engine (SSME) and a two-dimensional impeller. Unsteady impeller blade pressure measurements were made for a second two-dimensional impeller with blade number and geometry identical to the two-dimensional impeller used for the diffuser vane pressure measurements. The experiments were conducted with different flow coefficients and different radial gaps between the blade trailing edge and the diffuser vane leading edge (5% and 8% of the impeller discharge radius for the two-dimensional impellers, and 1.5% and 4.5% for the impeller of the HPOTP). The largest pressure fluctuations on the diffuser vanes and the impeller blades were found to be of the same order of magnitude as the total pressure rise across the pump. On the diffuser vanes, the largest pressure fluctuations were observed on the suction side of the vane near the leading edge, whereas on the impeller blades the largest fluctuations occurred at the blade trailing edge. The resulting lift on the diffuser vane was computed from the pressure measurements; the magnitude of the fluctuating lift was found to be larger than the steady lift.

Introduction

Blade and vane design in diffuser pumps is currently based upon the assumption that the flow in both the impeller and the diffuser is steady. This, however, implies that the radial gap between the impeller discharge and the diffuser inlet is large so that no flow unsteadiness of any kind due to rotor/stator interaction occurs. If, however, the radial gap between impeller blades and diffuser vanes is small, i.e., of the order of a small percentage of the impeller discharge radius, as is actually the case for many diffuser pumps, there are strong interactions which influence both the aerodynamic and the structural performance of the impeller blades and the diffuser vanes.

Most of the experimental work on unsteady blade pressures due to rotor/stator interaction has been done in axial turbomachinery. Among others, Dring (1982) investigated the blade row interaction in an axial turbine, and Gallus reported measurements for axial compressors (1979 and 1980). In radial turbomachinery, impeller blade pressure measurements were reported by Iino (1985). The radial gap between impeller blades and diffuser vanes was small, so that significant pressure fluctuations on the impeller blades were observed. Furthermore, it was found that blade and vane angle may have an important influence on the blade pressure fluctuations.

Herein, results of an investigation of rotor/stator interaction in centrifugal pumps with radial diffusers will be reported. Steady and unsteady diffuser vane pressure measurements were made for two impellers, a two-dimensional impeller, referred to as Impeller Z1, and one half of the double suction pump of the High Pressure Oxygen Turbopump (HPOTP) of the Space Shuttle Main Engine (SSME), referred to as Impeller R. A second two-dimensional impeller, referred to as Impeller Z2, with the same blade number and the same blade geometry as Impeller Z1, was used for impeller blade pressure measurements. For those measurements different diffuser vane configurations were used to investigate the influence of the vane number and the vane leading edge mean line angle (also referred to as vane angle) on the impeller blade pressure fluctuations. Steady and unsteady computations of the lift on the diffuser vanes were made from the vane pressure measurements. Results of these measurements have previously been reported by Arndt (1988a and 1988b).

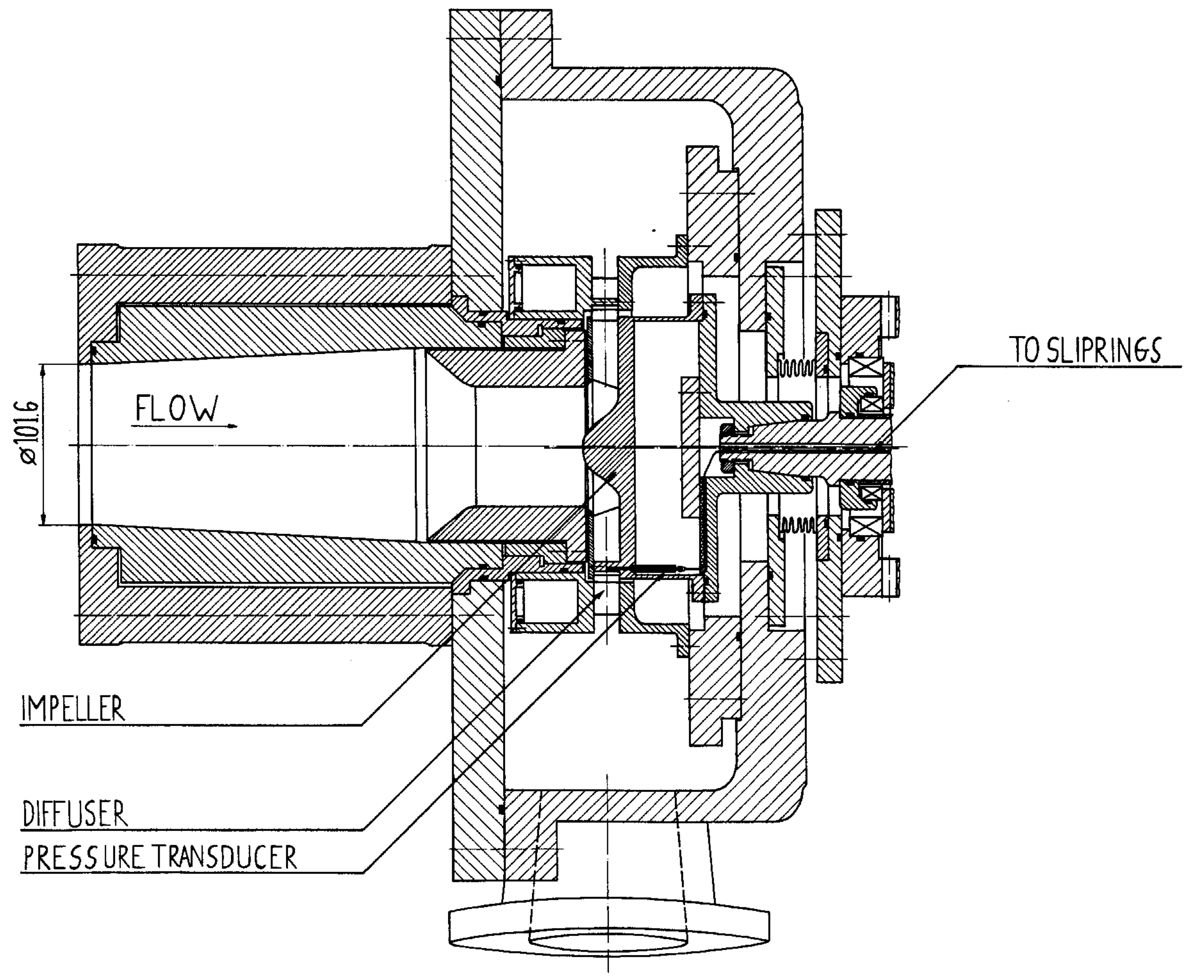


Fig. 1 Simplified view of the test section.

During the test, the impellers could only be positioned at locations on an orbit concentric to the diffuser center (orbit radius=1.27 mm), so that the radial gap between the impeller blade trailing edge and the leading edge of any particular instrumented diffuser vane could be varied between 1.5% and 4.5% of the impeller discharge radius for Impeller R, and 5% and 8% of the impeller discharge radius for the two–dimensional impellers. Similarly, for the impeller blade pressure measurements, the radial gap between the instrumented impeller blade and the diffuser vanes varied between 5% and 8% of the impeller discharge radius during one impeller revolution. Are the blade and vane pressure fluctuations measured for the "local" radial gaps between impeller blades and diffuser vanes representative for diffuser pumps in which the radial gap between the impeller blades and the diffuser vanes is uniform? During one shaft revolution, the impeller flow is subjected to disturbances occurring at two different frequencies (low frequency disturbances such as rotating stall etc. are excluded from this discussion since the present investigation did not focus on such phenomena); namely a disturbance due to the presence of the diffuser vanes occurring at vane passage frequency, f_v, and a disturbance due to the varying radial gap between impeller blades and diffuser vanes during one shaft revolution occurring at shaft frequency, f_s. For the centrifugal pump stages investigated, the vane passage frequency was an order of magnitude larger than the shaft frequency ($6 \leq (f_v/f_s) \leq 12$). Hence, it is inferred that the flow about an impeller blade passing a diffuser vane at a certain radial gap can be considered "quasiperiodic". That is to say the pressure fluctuations experienced by that particular impeller blade and that particular diffuser vane are representative of the pressure fluctuations the impeller blades and the diffuser vanes experience in diffuser pumps with a uniform radial gap. The proximity of neighboring diffuser vanes (for the instrumented impeller blade) and neighboring impeller blades (for the instrumented diffuser vane) with a slightly different radial gap is considered a small perturbation with only a small effect on the blade and vane pressure fluctuations measured on the particular instrumented impeller blade and diffuser vane.

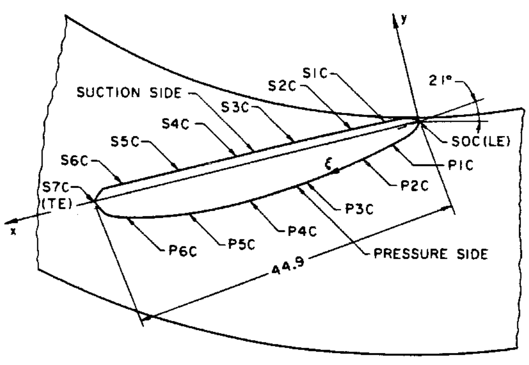


Fig. 2 Diffuser vane showing the location of the pressure taps at mid vane height.

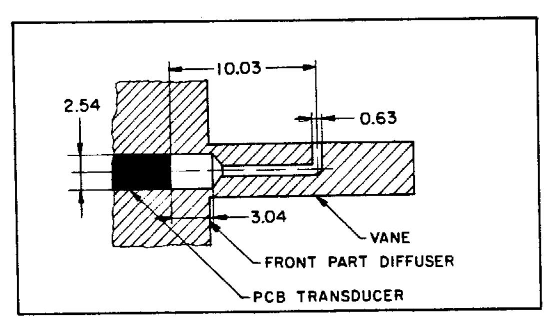


Fig. 3 Geometry of the vane and blade pressure taps.

Test Facility and Instrumentation

The experiments were conducted in a recirculating water test loop. A simplified view of the test section is shown in figure 1 with the two-dimensional impeller for the blade pressure measurements installed. Geometrical data for the three impellers used during the tests are given in Table I.

Table I: Impeller Geometry.

Impeller	z_b	α^*	t_b	d_2	b_2
Impeller R	8*	$\approx 36°$	15.00 mm	167.64 mm	14.48 mm
Impeller Z1	5	25°	13.00 mm	161.92 mm	15.75 mm
Impeller Z2	8	25°	13.00 mm	161.92 mm	15.75 mm

* Impeller R has eight blades in total, four full and four partial blades, and an inducer with four inducer blades. The impeller discharge impeller had to be reduced from 172.7 mm to 167.64 mm for the impeller to fit into the test facility.

The vaned diffuser used for the diffuser vane pressure measurements, referred to as Diffuser S, is a straight wall constant width diffuser with nine vanes and no volute. The flow is discharged from the diffuser into a large housing. The shape of a vane, with the pressure taps at mid vane height, is shown in figure 2. It is identical to the one used in an early version of the diffuser of the HPOTP of the SSME; however, the number of vanes was reduced from seventeen in the diffuser of the SSME to nine in the experiment. A second diffuser, of identical side wall geometry and diffuser channel width as Diffuser S, but permitting variable diffuser vane configurations was used to investigate the effects of the number of diffuser vanes and of the vane leading edge mean line angle (also referred to as the vane angle), β^*, on the impeller blade pressure measurements. This employed circular arc vanes with the geometry listed in Table II.

Table II: Geometry of the Circular Arc Vanes.

R_{mean}	88.91 mm
c	56.90 mm
t_v	4.19 mm

Four different diffusers were used for the unsteady impeller blade pressure measurements (diffuser vane pressure measurements could only be made using Diffuser S); the details are listed in Table III.

Table III: Diffuser Geometry.

Diffuser	Vane Type	z_v	β^*	d_3	b_3
Diff. F	circ. arc	12	20°	172.72 mm	14.99 mm
Diff. G	circ. arc	6	20°	172.72 mm	14.99 mm
Diff. H	circ. arc	6	10°	172.72 mm	14.99 mm
Diff. S	see fig. 2	9	21°	172.72 mm	14.99 mm

Steady vane pressure measurements were obtained by using conventional mercury manometers. The wall pressure at the intake about 260 mm upstream of the inducer blade leading edge of Impeller R was used as a reference pressure. The experimental error on those measurements was estimated to be $\pm 0.5\%$.

Piezoelectric pressure transducers from PCB Inc. were used for the unsteady measurements. The eigenfrequency (resonant frequency) of the transducer in air was listed as 300 kHz. The linearity of the calibration provided by the manufacturer was within 2%. The eigenfrequency of the pressure tap, the geometry of which is shown in figure 3, was estimated to be approximately 8000 Hz. The spectrum of unsteady vane pressure measurements (figure 4) taken at suction side tap S2C at 1800 rpm and $\phi = 0.12$ shows that the estimate of the eigenfrequency was reasonable. It can be seen that the blade passages frequency and its higher harmonics are "far" removed from the eigenfrequency of the tap, so that amplification and phase shift of the signal were negligible.

The data were sampled and discretized in a 16 channel data acquisition system. An encoder on the main shaft was used to trigger the data taker and to provide a clock for the data taker. 1024 data per main shaft cycle were taken for shaft speeds up to 1800 rpm, 512 for higher shaft speeds. Since the signal contained some noise, the unsteady data were ensemble averaged over one impeller revolution for the two–dimensional impellers and over two impeller blade passages, a full and a partial blade passage, for Impeller R. The experimental error was found to be less than ±5% for the magnitude and less than ± 2 degrees (360 degrees corresponding to one impeller blade passage) for the phase of the ensemble averaged unsteady pressure measurements.

Vane Pressure Measurements for Impeller R

Unsteady Vane Pressure Measurements. The unsteady vane pressure measurements are presented as an unsteady vane pressure coefficient, normalized by the dynamic pressure based on impeller tip speed. In figure 5 a sample of unsteady vane pressure measurements taken at pressure tap S2C for the best efficiency flow coefficient, $\phi = 0.12$, and R_3/R_2=1.015 is shown. It is evident that the pressure fluctuations are periodic with impeller blade passage (c.f. the spectrum of this sample of unsteady data presented in figure 4) and of the same order of magnitude as the total pressure rise across the pump (c.f. figure 6a). The lowest pressure occurs after the impeller blade suction side has passed the diffuser vane leading edge. Again, the noise in the measurement is due to the resonant frequency of the pressure tap.

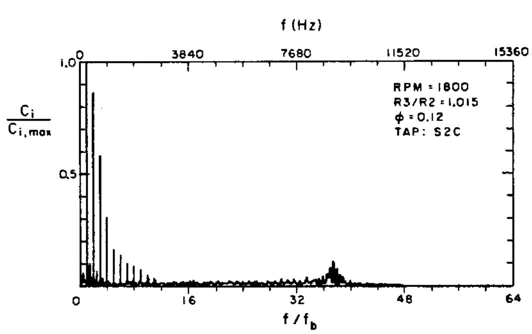


Fig. 4 Spectrum of unsteady diffuser vane pressure measurements at pressure tap S2C for Impeller R ($\phi = 0.12, R_3/R_2 = 1.015$).

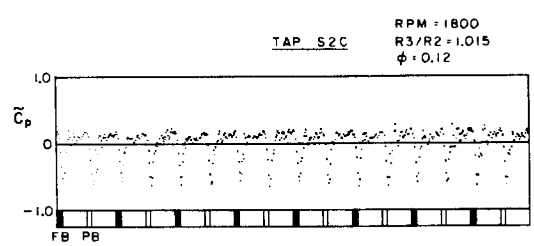


Fig. 5 Unsteady diffuser vane pressure measurements at pressure tap S2C for Impeller R.

Magnitude of Vane Pressure Fluctuations. In figure 7 data on the magnitude of the ensemble averaged unsteady vane pressure fluctuations at mid vane height are presented. The fluctuations are defined as the difference between the maximum and minimum pressure in the averaging period which corresponds to one full and one partial impeller blade passage. It can be seen that the largest fluctuations occur on the front half of the vane suction side. Those fluctuations are of the same order of magnitude as the total pressure gain across the pump. Furthermore, the fluctuations on the pressure side are significantly smaller than those on the suction side. Increasing the radial gap from 1.5% to 4.5% resulted in a significant decrease, of about 50%, in the large fluctuations on the front half of the vane suction side and at the pressure tap on the pressure side closest to the leading edge. At most other measurement locations the fluctuations decreased slightly or remained constant. Only at one pressure tap, P5C, on the rear half of the pressure side, did they increase with increasing radial gap. It is interesting to notice, that although the fluctuations decrease with distance along both the pressure and suction sides there is a small increase as the trailing edge is approached. To investigate the dependence of the magnitude of the fluctuations on the flow coefficient, measurements were obtained for radial gaps of 1.5% and 4.5%, for a total of eleven flow coefficients ranging from $\phi = 0.05$ to $\phi = 0.15$, at one particular pressure tap on the vane suction side close to the vane leading edge (Tap S2C). In figure 8, the magnitude of the fluctuations as a function of flow coefficient and radial gap, relative to the magnitude of the fluctuations for maximum flow, $\phi = 0.15$, at a radial gap of 1.5%, is presented. For both radial gaps, the fluctuations were observed to be largest for the maximum flow coefficient. Increasing the radial gap from 1.5% to 4.5% resulted in an approximately 50% decrease of the fluctuations. The fluctuations attain relative minima and maxima for $\phi = 0.09$ and $\phi = 0.07$ (for a radial gap of 1.5%), and for $\phi = 0.11$ and $\phi = 0.07$ (for a radial gap of 4.5%).

Steady and Unsteady Lift Computations. From the vane pressure measurements described earlier, the force on the vane at mid vane height was computed. The steady force was computed from the steady pressure distribution around the vane. Superimposing the steady and ensemble averaged unsteady pressure measurements, the ensemble averaged vane pressure distribution was obtained. From the ensemble averaged pressure distribution, the ensemble averaged force was computed using third order periodic splines through the measured pressure values. A periodic spline fit was chosen to ensure continuity for the pressure and the first two pressure derivatives at the leading and trailing edges. The steady and ensemble averaged forces were computed from

$$\overline{\mathbf{F}} = -\oint (\overline{p}_v - \overline{p}_{up})(\xi)\mathbf{n}d\xi \quad \text{and} \quad \mathbf{F_{av}} = -\oint (\overline{p}_v + \tilde{p}_{v,av} - \overline{p}_{up})(\xi)\mathbf{n}d\xi.$$

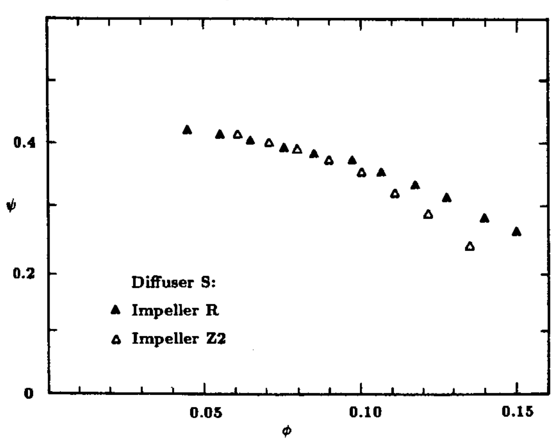


Fig. 6a Performance curves for Impeller R and Diffuser S and Impeller Z2 and Diffuser S.

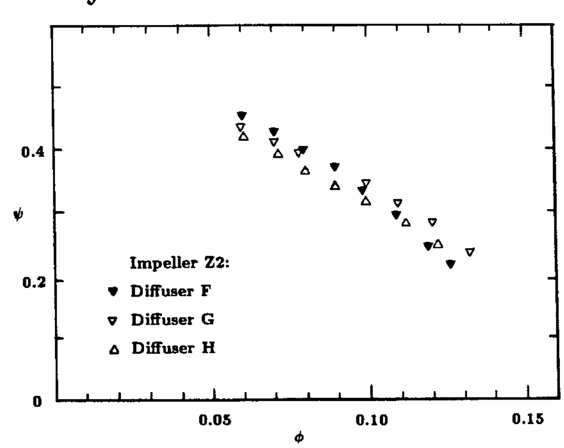


Fig. 6b Performance curves for Impeller Z2 and Diffusers F, G and H.

The lift on the vane was defined as the component of the force on the vane normal to the chord joining the vane leading and trailing edges. It was defined positive in positive y direction shown in figure 2. Normalized by the dynamic pressure based on impeller tip speed and on the vane chord, the ensemble averaged lift will be presented as an ensemble averaged lift coefficient.

In figure 9 the ensemble averaged vane pressure distribution at mid vane height is shown for two different positions of the impeller blades relative to the instrumented vane for best efficiency flow coefficient, $\phi = 0.12$, at 1800 rpm, and a radial gap of 1.5%. As the impeller blade passes the diffuser vane, the vane pressure on the vane suction side drops below upstream pressure, resulting in a large pressure difference between the vane suction and the vane pressure side. When the impeller blade is approximately halfway between two adjacent diffuser vanes those large pressure differences are greatly reduced.

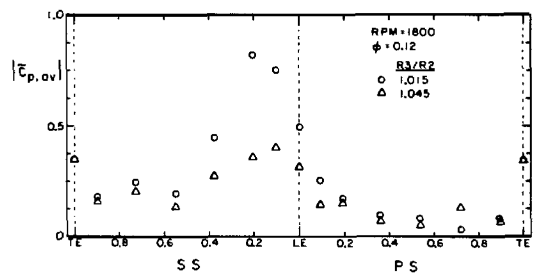
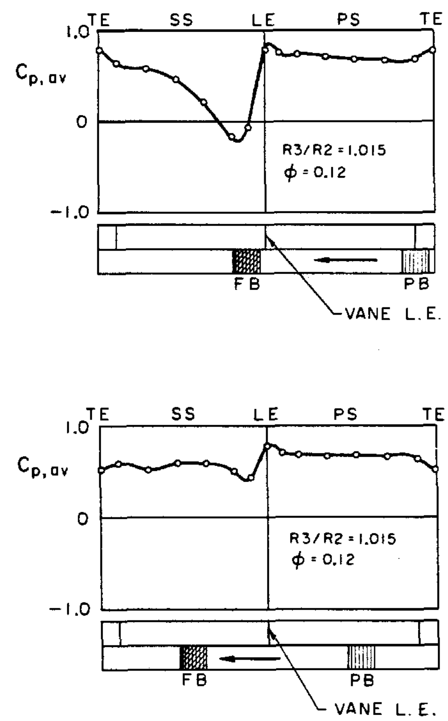


Fig. 7 Magnitude of ensemble averaged pressure fluctuations at mid vane height for Impeller R ($\phi = 0.12, R_3/R_2 = 1.015$ and 1.045).



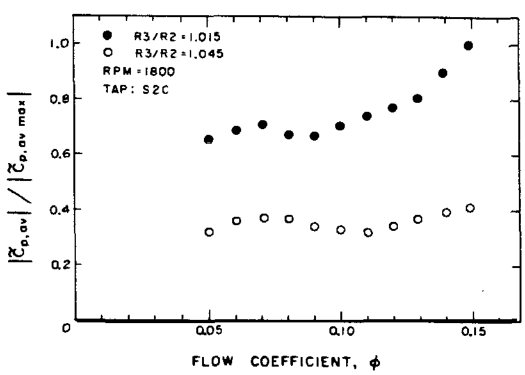


Fig. 8 Magnitude of ensemble averaged vane pressure fluctuations at pressure tap S2C for Impeller R ($R_3/R_2 = 1.015$ and $1.045, \phi = 0.05 - 0.15$). Best efficiency flow coefficient: $\phi = 0.12$.

Fig. 9 Ensemble averaged pressure distribution at vane mid height for Impeller R ($\phi = 0.12, R_3/R_2 = 1.015$).

In figure 10 the ensemble averaged lift on the vane is presented for $\phi = 0.15, 0.12$, and 0.09 for a radial gap 4.5%. The fluctuations are largest for $\phi = 0.15$ and decrease for $\phi = 0.12$ and $\phi = 0.09$. Differences in the magnitude of the lift fluctuations occur depending on whether a partial or a full impeller blade has passed the diffuser vane. For best efficiency point and just below ($\phi = 0.12$ and $\phi = 0.09$), the lift fluctuations were larger for a full impeller blade passing the vane than for a partial impeller blade (by 5% and 22%, respectively). For maximum flow($\phi = 0.15$), however, the lift fluctuations were about 8% larger for a partial impeller blade than for a full impeller blade. The fluctuating lift was found, depending upon flow coefficient, to be up to three times larger than the steady lift (Arndt, 1988a and 1988b). The observations that the fluctuating lift decreases with decreasing flow coefficient is different from that reported by Gallus (1979 and 1980) who observed that the fluctuating lift on the stator blades was smallest for maximum flow, and increased with decreasing flow coefficient.

Vane Pressure Measurements for Impeller Z1

Comparing the Vane Pressure Measurements for Impeller R and Impeller Z1. The magnitude of the ensemble averaged vane pressure fluctuations for Impeller R and Impeller Z1 are presented in figure 11. The measurements were made for comparable flow coefficients ($\phi = 0.12$ for Impeller R and $\phi = 0.10$ for Impeller Z1) and for comparable radial gaps between the respective impeller blade trailing edges and the instrumented vanes (4.5% of the impeller discharge radius for Impeller R and 5.0% of the impeller discharge radius for Impeller Z1). The measurements were made at different shaft speeds, 3000 rpm for Impeller Z1 and 1800 rpm for Impeller R. This, however, should have no influence on comparing the sets of data, for the vane pressure fluctuations if normalized by the dynamic head based on impeller tip speed were found to be virtually independent of shaft speed for both impellers. It can be seen that with the exception of one pressure tap the pressure fluctuations on the front half of the diffuser vane (pressure taps S0C–S4C and P1C–P4C) are of nearly identical magnitude. At pressure tap S1C (the pressure tap on the suction side closest to the vane leading edge), the fluctuations for Impeller R are about twice as large as those for Impeller Z1. The magnitude of the normalized lift fluctuations resulting from the unsteady pressure distribution on the diffuser vane is approximately equal for both Impeller Z1 and Impeller R, $|\tilde{c}_{L,av}| \approx 0.2$ (Arndt 1988a and 1988b).

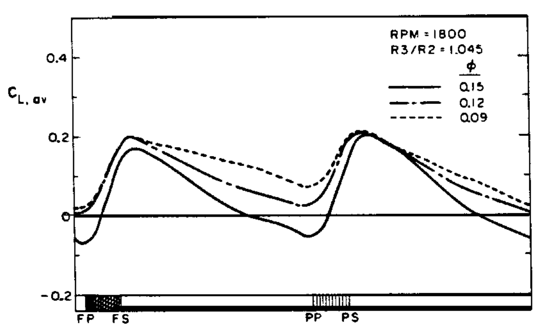


Fig. 10 Ensemble averaged lift on the diffuser vane at mid vane height for Impeller R ($\phi = 0.15, 0.12$ and $0.09, R_3/R_2 = 1.045$).

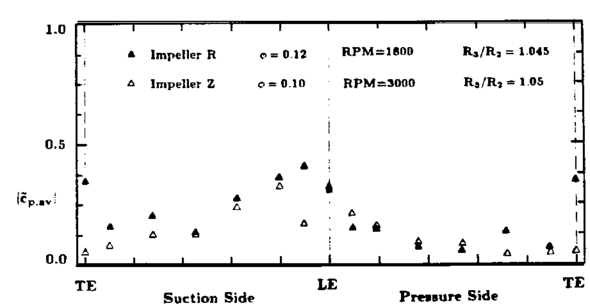


Fig. 11 Magnitude of ensemble averaged pressure fluctuations at mid vane height for Impeller R ($\phi = 0.12, R_3/R_2 = 1.045$) and Impeller Z1 ($\phi = 0.10, R_3/R_2 = 1.05$).

As for Impeller R, for one pressure tap, S2C, unsteady vane pressure measurements were made for Impeller Z1 over a range of flow coefficients, from the maximum flow coefficient, $\phi = 0.135$, to $\phi = 0.06$. In figure 12, the magnitude of the fluctuations as functions of flow coefficient and radial gap, relative to the magnitude of the fluctuations for maximum flow, $\phi = 0.135$, at a radial gap of 5%, are presented. It can be seen that the fluctuations are largest for maximum flow and decrease with decreasing flow coefficient. Increasing the radial gap between the impeller blades and the diffuser vanes resulted in a decrease of about 50% in the magnitude of the fluctuations.

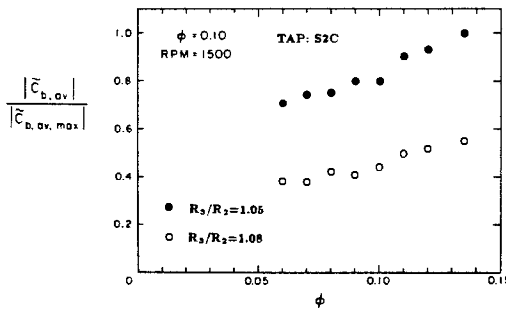


Fig. 12 Magnitude of ensemble averaged vane pressure fluctuations at pressure tap S2C for Impeller Z1 ($R_3/R_2 = 1.05$ and $1.08, \phi = 0.05$–0.135).

Unsteady Impeller Blade Pressure Measurements for Diffuser S

Unsteady pressure measurements were made at three pressure taps on the blades of Impeller Z2. The three pressure taps were located at the blade pressure side ($r/R_2 = 0.987$), at the blade suction side ($r/R_2 = 0.937$), and at the blade trailing edge ($r/R_2 = 1.00$). Here, the measurements made for Diffuser S and the two–dimensional impeller are presented. Since the impeller was positioned eccentrically to the diffuser center, the radial gap between impeller blade trailing edge and diffuser vane leading edge varied during one impeller revolution from 5% to 8%. For the smallest radial gap, $R_3/R_2 = 1.05$, the magnitudes of the ensemble averaged blade pressure fluctuations are presented in figure 13. The measurements were made for a total of eight flow coefficients, ranging from maximum flow, $\phi = 0.135$, to $\phi = 0.06$.

It can be seen that the magnitudes of the pressure fluctuations range from the same order of magnitude as the total pressure rise across the pump at the trailing edge tap, to an order of magnitude smaller than the total pressure rise across the pump at the suction side pressure tap.

The largest fluctuations occur at the trailing edge. They increase significantly with the flow coefficient decreasing from the maximum flow coefficient, $\phi_{max} = 0.135$, to the smallest flow coefficient investigated, $\phi = 0.06$. The pressure fluctuations at the blade pressure and blade suction side pressure taps do not change very substantially with flow coefficient. Maxima are attained for $\phi = 0.11$ on the pressure surface, for $\phi = 0.10$ on the suction surface, and for $\phi = 0.06$ on the suction and pressure surfaces. Minima are attained for maximum flow, $\phi = 0.135$, on both the pressure and suction surfaces, and for $\phi = 0.09$ on the pressure surface, and for $\phi = 0.08$ on the suction surface. On the pressure surface, the fluctuations were about two to three times larger than on the suction surface At maximum flow, the pressure surface fluctuations were even slightly larger than those at the trailing edge. Not increasing significantly with decreasing flow coefficient, however, the fluctuations on the pressure surface for low flow coefficients were only about half as large as those at the trailing edge.

Unsteady Impeller Blade Pressure Measurements for Different Diffusers

Unsteady impeller blade pressure measurements were also made for a second diffuser, of identical sidewall geometry as Diffuser S, but permitting variable diffuser vane configurations to investigate the effects of the number of diffuser vanes and the diffuser vane angle on the impeller blade pressure fluctuations. Three different vane configurations employing circular arc vanes were tested. Diffuser F employed twelve vanes with a vane angle, β^*, of twenty degrees, Diffuser G six vanes with vane angle of 20 degrees, and Diffuser H six vanes with a vane angle of 10 degrees. Performance curves for the three diffusers and Impeller Z2 are presented in figure 6b. The influence of the different diffusers on the unsteady pressure measurements at any of the three pressure taps are presented in figures 14, 15 and 16 (in figure 14 for the trailing edge pressure tap, in figure 15 for the pressure surface tap, and in figure 16 for the suction surface tap). In each figure, the magnitude of the pressure fluctuations for the smallest radial gap between the instrumented impeller blade blade and the diffuser vanes, $R_3/R_2 = 1.05$, is presented versus flow coefficient.

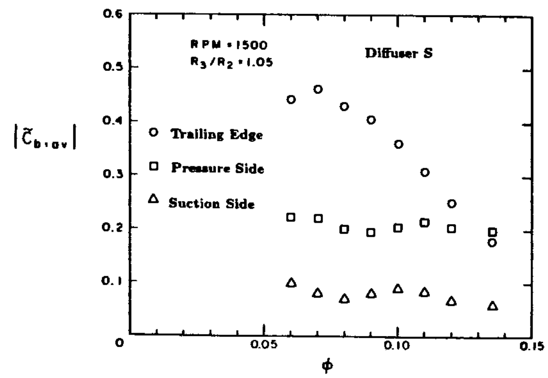


Fig. 13 Magnitude of ensemble averaged blade pressure fluctuations for Diffuser S ($\phi = 0.06 - 0.135, R_3/R_2 = 1.05$).

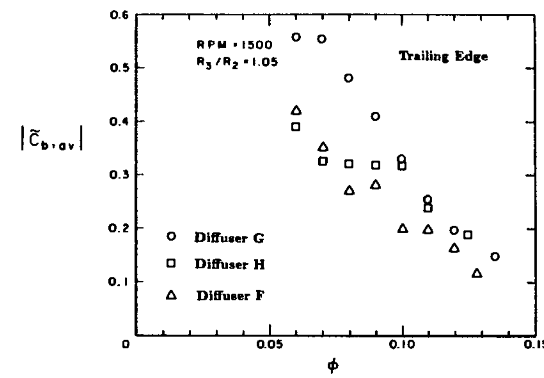


Fig. 14 Magnitude of ensemble averaged blade pressure fluctuations at the impeller blade trailing edge for Diffusers F, G and H ($\phi = 0.06 - 0.135, R_3/R_2 = 1.05$).

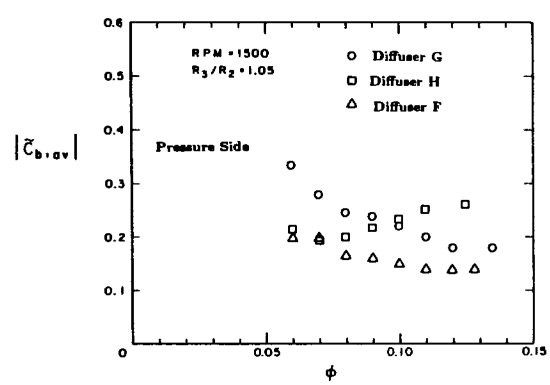


Fig. 15 Magnitude of ensemble averaged blade pressure fluctuations at the impeller blade pressure side pressure tap for Diffusers F, G and H ($\phi = 0.06 - 0.135, R_3/R_2 = 1.05$).

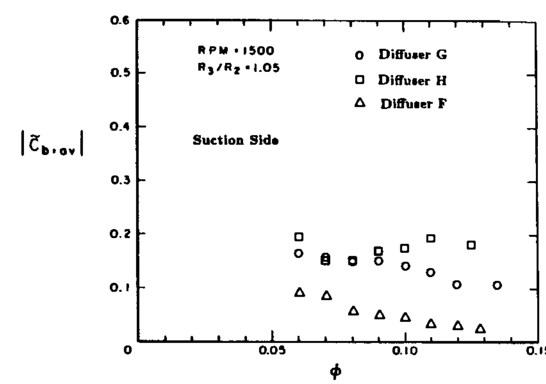


Fig. 16 Magnitude of ensemble averaged blade pressure fluctuations at the impeller blade suction side pressure tap for Diffusers F, G and H ($\phi = 0.06 - 0.135, R_3/R_2 = 1.05$).

From those three figures, it can be seen that vane number and vane angle do have a most significant influence on the impeller blade pressure fluctuations. The increase in vane number from six to twelve at a fixed vane angle of 20 degrees resulted in a decrease of the blade pressure fluctuations at all blade pressure taps and for all flow coefficients. The reduction varied, depending upon flow coefficient and pressure tap, between 15% and 70%. Decreasing the vane angle from 20 degrees to 10 degrees for a constant number of diffuser vanes ($z_v = 6$) resulted in an increase of the pressure fluctuations for the large flow coefficients of up to 75%, whereas for low flow coefficients the fluctuations decreased by up to 30%. For the trailing edge tap, the magnitude of the pressure fluctuations remained unchanged for large flow coefficients, and decreased for low flow coefficients.

Conclusion

Steady and unsteady diffuser vane pressure measurements, and unsteady impeller blade pressure measurements in diffuser pumps were made with the objective of investigating rotor/stator interaction. Measurements were presented for radial gaps between the impeller discharge and the diffuser inlet ranging from 1.5% to 5% of the impeller discharge radius. The diffuser vane pressure fluctuations were found to be largest on the vane pressure side near the vane leading edge, and were of the same order of magnitude as the total pressure rise across the pump, They were largest for the maximum flow coefficient, and decreased with decreasing flow coefficient. The largest impeller blade pressure fluctuations (also of the same order of magnitude as the total pressure rise across the pump) occurred at the impeller blade trailing edge. In contrast to the large pressure fluctuations at the diffuser vane suction side, the fluctuations at the impeller blade trailing edge were smallest for the maximum flow coefficient, and increased with decreasing flow coefficient. The number of diffuser vanes and the diffuser vane leading edge mean line angle were found to have a significant influence upon the impeller blade pressure fluctuations. The lift on the diffuser vane, steady and unsteady, was computed from the vane pressure measurements. The magnitude of the fluctuating lift was found to be substantially larger than the steady lift.

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Nomenclature

A_2	impeller discharge area, $A_2 = 2\pi R_2 b_2$
b_2, b_3	impeller discharge width, diffuser inlet width
c	vane chord
c_b	blade pressure coefficient, $c_b = p_b/(1/2)\rho u_2^2$
c_i	magnitude of i'th Fourier coefficient, normalized by $(1/2)\rho u_2^2$
c_L	lift coefficient, $c_L = L/(1/2)\rho u_2^2 c$
$\tilde{c}_p$	unsteady vane pressure coefficient, $\tilde{c}_p = \tilde{p}_v/(1/2)\rho u_2^2$
$\tilde{c}_{p,av}$	ensemble averaged unsteady vane pressure coefficient, $\tilde{c}_{p,av} = \tilde{p}_{v,av}/(1/2)\rho u_2^2$
$c_{p,av}$	ensemble averaged vane pressure coefficient, $c_{p,av} = (\overline{p}_v - \overline{p}_{up} + \tilde{p}_{v,av})/(1/2)\rho u_2^2$
$\mathbf{F}$	force vector on diffuser vane
f	frequency
L	lift (= component of the force vector on the vane normal to the chord joining the vane leading and the vane trailing edge)
$\mathbf{n}$	outward normal vector on diffuser vane
p, p_t	pressure, absolute pressure
Q	flow rate
r	radius
R_2, R_3	impeller discharge radius, diffuser inlet radius
R_{mean}	mean line radius of the circular arc vanes
rpm	revolutions per minute
t_b, t_v	impeller blade thickness, diffuser vane thickness
u_2	impeller tip speed, $u_2 = 2\pi R_2(rpm/60)$
x,y	diffuser vane coordinates
z_b, z_v	number of impeller blades, number of diffuser vanes
α^*	impeller blade trailing edge angle (= impeller blade angle)
β^*	diffuser vane leading edge mean line angle (= diffuser vane angle)
ξ	parametric diffuser vane coordinate
ρ	density
ϕ	flow coefficient, $\phi = Q/u_2 A_2$
ψ	total head coefficient, $\psi = (p_{down} - p_{up})/\rho u_2^2$

Subscripts

av	ensemble averaged
b	impeller blade
$down, up$	downstream, upstream
max	maximum
s	shaft
v	diffuser vane

Superscripts

$\bar{}$	steady
$\tilde{}$	unsteady

Abbreviations

FB,PB	full impeller blade, partial impeller blade
FP,FS	pressure side of full impeller blade, suction side of full impeller blade
HPOTP	High Pressure Oxygen Turbopump
LE,TE	diffuser vane leading edge, diffuser vane trailing edge
PCB	PCB Piezoelectronics, INC. Depew, NY 14043
PP,PS	pressure side of partial impeller blade, suction side of full impeller blade
PS,SS	diffuser vane pressure side, diffuser vane suction side